International Journal of Mechanical and Production
Engineering Research and Development (IJMPERD)
ISSN (P): 2249-6890; ISSN (E): 2249-8001
Vol. 8, Issue 4, Aug 2018, 791-802

CAM PROFILE OPTIMISATION OF SINGLE CYLINDER DIESEL ENGINE TO REDUCE NOISE AND VIBRATION

VISHALKUMAR K. DHUMMANSURE[1], D. RAMESH RAO[2] & YOGESH BAGMAR[3]

[1]*Research scholar, Department of Mechanical Engineering, Bapuji Institute of Engineering and Technology, Davangere, Karnataka, India*

[2]*Professor, Department of Mechanical Engineering, Presidency University, Bengaluru, Karnataka, India*

[3]*Research Scholar, Department of Mechanical Engineering, Sinhgad Academy of Engineering, Kondhwa, Pune, Maharashtra, India*

ABSTRACT

The optimization of the cam profile plays an important role to reduce noise and vibration in an internal combustion engine. This study analyzes a method for the design and experimentation of different cam profiles. The proposed method commences with the use of higher order polynomials for the cam profile. Subsequently, an optimum cam profile is redesigned and used in the engine. The improvement of the cam profile is based on optimized factors such as velocity and acceleration in the cam profile. The vibration and noise due to impact between cam and tappet can be controlled by adjusting the ramp height and the velocity in the ramp area. Also, there is a significant improvement in Sound quality. The experimental result shows a reduction in overall noise by 1.86dB (A) in case of optimized cycloidal cam profile of single cylinder engine.

KEYWORDS: Cam Profile, Ramp Area, Sound Quality, Valve Train Assembly & Vibration Response

Received: Jun 15, 2018; **Accepted**: Jul 05, 2018; **Published**: Jul 30, 2018; **Paper Id**.: IJMPERDAUG201882

Original Article

1. INTRODUCTION

The noise of the valve train system is one of the most important parameters during engine operation. It is considered as unpleasant. Study of cam profile can be one of the parameters to reduce the radiated noise level of the engine. Cam profile can be optimized mathematically in many ways to reduce velocity and acceleration of follower. Using the splined curve is one of the best methods to have control on the motion [1, 2]. Various parameters influencing (pressure angle, undercutting) are considered while designing the cam profile are studied.

The design and analysis of cam profile were carried out using three circular arc profiles [3]. The equations for a geometric model of the cam are derived and investigated numerically. Design of a constant force mechanism by preloading linear spring was accomplished [4]. Bezier curves have been used in cam profile design to maximize the area of stiffness-torque curves and on the other hand, its feasibility to operate as a knee joint [5]. A differential evolution technique was used for different configurations with different constraints and cost functions. The author suggested the best cam shape for knee joint application but depending on the particular application there is always an optimal solution.

A numerical method was proposed to optimize the cam profile where the space is restricted and forces are high [6]. Higher order curves of the order 6 and 8 can be used for the design of cam profiles but on the other hand, continuity has to be ensured for second and third derivatives. Parametric equation was derived and analyzed by math method [7]. Improvement was achieved by design of fuel supply cam profile with constant pressure injection. Author concludes that the injection requirements can be satisfied by designing a cam profile using parametric equations and further the equations simplify the design process. A constant pressure cam was designed to improve the performance of the engine [8]. The model was based on three constraints; 1) pump pressure equals to nozzle pressure, 2) cam speed should decrease with the increase when injection duration increases and 3) cam acceleration gradient should be zero. Author concludes that by maintaining same peak pressure the engine runs at the higher speed as cam delivers more injection quantity.

Constant acceleration profiles have been analyzed along with the relationship between positive acceleration, negative acceleration and lift area [9]. The method can be applied to other cam profiles.

Bazier curves were used to design and optimize the cam profile for mandible of patients[10]. Author concludes that accurate reproduction of 3D model is possible by representing the cam profile by curve equations. The dynamic behavior of flexible components (such as valve springs) is studied by developing kineto-elasto dynamics method. The effects of deflection and torsional vibration of the camshaft are also considered in the study [11]. The inertia toque of camshaft is one of the key factor influencing the vibration of the engine. The fluctuations in inertia torque can reduced by designing a balancing mechanism to isolate the vibration camshaft [12, 13]. The balancer cam not only reduces the inertia torque, also eliminates the preload used to avoid cam jump phenomena.

A Lot of study has been carried out to design the cam profile using higher order curves. The studies focused only on motion. However, the effect on Noise and vibration was considered a little. This paper proposes a methodology to design the cam profile using higher order polynomials to reduce the noise and vibration level of the engine.

2. OBJECTIVES

- To measure the noise radiated and vibration response of existing valve train assembly.
- To design the cam profile using higher order polynomials.
- To manufacture the camshaft with higher order polynomials.
- To measure the Noise radiated and vibration levels in modified cam assembly.

Cam assembly of a single cylinder 8 HP air cooled (specifications are given in table1) is considered for the study. A tangent cam profile is used for valve train system. The engine is the self-governed type with a speed of 1300 rpm at full load and 1400 rpm at no load condition. The objective of this study is to investigate the influence of cam profile on vibration response and noise radiated from the valve train assembly. A methodology of the proposed research is given in figure1.

Table 1: Engine Specification

Engine Capacity	553 cc, Single Cylinder
Speed	2600 rpm
Power at max speed	8 HP
Cam shaft speed	1300 rpm
Valve train	Mechanical push rod type

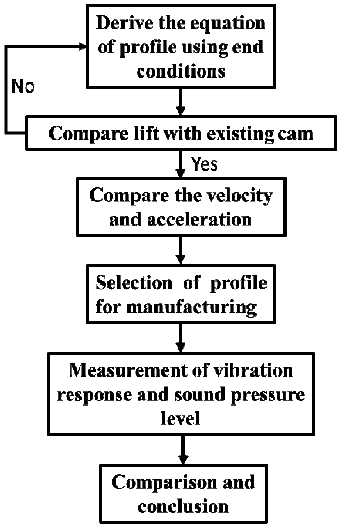


Figure 1: Methodology of the Proposed Work

3. THEORETICAL BACKGROUND

In the valve train system cam is the most important element which is responsible to excite the motion in the system. In this paper, the cam profile design for high-speed engine is the main task for vibration and ultimately noise reduction of valve train system.

The main parameters for a cam profile design are

- Open/Close Ramp area- This is the part of a cam profile where the follower motion starts/ends. It is the main region where a cam designer has to concentrate because it is the main source of impact noise in a system.
- Flank area- This is the area where the tappet gains the required acceleration which is important for the performance of the valves. The main task to design this region is to make sure that the jerk involved must be minimum.
- Nose- This is the higher lift area of the cam profile of a cam. The main objective to design this area is to avoid the heavy acceleration.

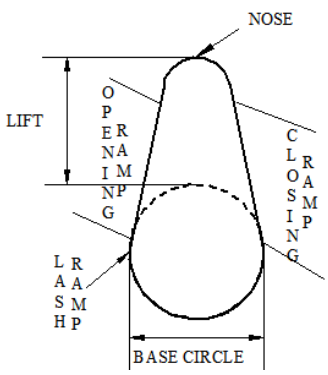


Figure 2: Cam Profile Terminology

3.1. Cycloidal Cam Profile

The equations relating the follower displacement, velocity and acceleration to cam rotation angle are as follows.

$$s = \frac{H}{\pi}\left[\frac{\pi\theta}{\beta} - \frac{1}{2}\sin\left(\frac{2\pi\theta}{\beta}\right)\right] \quad (1)$$

$$v = \frac{H\omega}{\pi\beta}\left[1 - \cos\left(\frac{2\pi\theta}{\beta}\right)\right] \quad (2)$$

$$a = \frac{2H\pi\omega^2}{\beta^2}\sin\left(\frac{2\pi\theta}{\beta}\right) \quad (3)$$

The general expression for a polynomial is given by

$$s= C_o+C_1q+C_2q^2+C_3q^3+\ldots\ldots +C_nq^n \quad (4)$$

s = displacement of follower

q = cam rotation angle

C_i = Constants (i = 0, 1 . . . n)

n = order of the polynomial.

For a polynomial cam of order n, there are n + 1 unknown constant coefficients. The values of constant can be determined by considering end conditions. These constants must satisfy the continuity condition in displacement, velocity and acceleration.

3.2. 3-4-5 Polynomial Cam

For a 5th order polynomial six boundary conditions are required (at initial position displacement, velocity and acceleration are zero. Also at the maximum lift position when cam has turned by rise angle, velocity, and acceleration are zero).

Simultaneous solution of equations yield:

$C_o = C_1 = C_2 = 0,\ C_3 = 10\frac{H}{\beta^3},\ C_4 = 15\frac{H}{\beta^4},\quad C_5 = 6\frac{H}{\beta^5}$ Where H is total lift and θ is angle of rise.

The equations for displacement, velocity and acceleration for 3-4-5 cam profile are as follows

$$s = H\left(\frac{\theta}{\beta}\right)^3\left[10 - 15\left(\frac{\theta}{\beta}\right) + 6\left(\frac{\theta}{\beta}\right)^2\right] \quad (5)$$

$$v = 30H\,\frac{\omega}{\beta}\left(\frac{\theta}{\beta}\right)^2\left[1 - 2\left(\frac{\theta}{\beta}\right) + \left(\frac{\theta}{\beta}\right)^2\right] \quad (6)$$

$$a = 60H\left(\frac{\omega}{\beta}\right)^2\left(\frac{\theta}{\beta}\right)\left[1 - 3\left(\frac{\theta}{\beta}\right) + 2\left(\frac{\theta}{\beta}\right)^2\right] \quad (7)$$

3.3. 4-5-6-7 Polynomial Cam

The third derivative of displacement to be zero at initial and total rise position one can have eight end conditions (jerk will be zero at initial and total rise position) and the polynomial will be 4-5-6-7 polynomial. Equations for displacement, velocity, and acceleration are derived as follows.

$$s = H\left(\frac{\theta}{\beta}\right)^4\left[35 - 84\left(\frac{\theta}{\beta}\right) + 70\left(\frac{\theta}{\beta}\right)^2 - 20\left(\frac{\theta}{\beta}\right)^3\right] \quad (8)$$

$$v = H\,\frac{\omega}{\beta}\left(\frac{\theta}{\beta}\right)^3 140\left[1 - 3\left(\frac{\theta}{\beta}\right) + 3\left(\frac{\theta}{\beta}\right)^2 - \left(\frac{\theta}{\beta}\right)^3\right] \tag{9}$$

$$a = H\left(\frac{\omega}{\beta}\right)^2\left(\frac{\theta}{\beta}\right)^2 420\left[1 - 4\left(\frac{\theta}{\beta}\right) + 5\left(\frac{\theta}{\beta}\right)^2 - 2\left(\frac{\theta}{\beta}\right)^3\right] \tag{10}$$

3.4. Splined Cam Profile

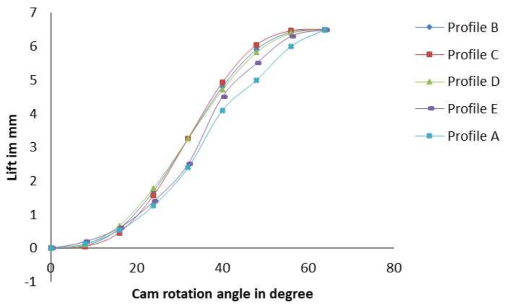
In the previous section, only one curve is used for the complete rise. This procedure does not give flexibility on the motion itself. In order to have control over the motion, the rise part can be divided into a number of parts. The endpoints of these parts are called knots. The first and last knot is the start and end of rise of the follower and the boundary conditions must be satisfied. At the interior knots, in order to have continuity, two adjacent polynomials must have the same value of slope and curvature. For each part, a polynomial of order n is written and knot coefficients are solved using boundary conditions.

3.5 Various Cam Profile Features

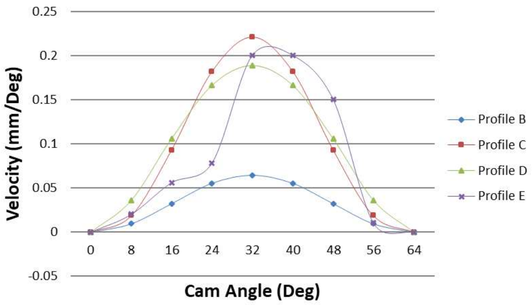
In the cam profile, the open Ramp area means the duration that the valve starts open from the base circle of the cam profile with the best-designed contact. A similar case is for close Ramp area where the valve starts to close. In the design process, lift curve is defined by considering engine performance first, and the next Ramp area is designed for smooth motion between cam and tappet. But as the open Ramp is a special area where cam and tappet are impacted physically and also during the closing of valve there is seat impact. For that Ramp is needed to optimize for noise reduction and improvement of sound quality. Details of various cam profiles designed are given in table 2.

Table 2: Cam Details

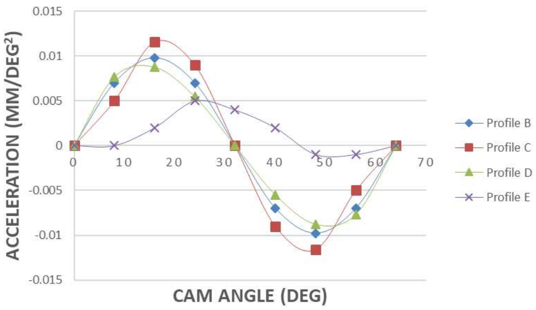
CAM Name	Lift (mm)	Period of lift Open/close (Degree)	Ramp Height (mm)	Ramp Velocity (mm/Degree)	Profile name
Profile A	6.5	64.3/64.3	0.2	-	Tangential Profile
Profile B	6.5	64.3/64.3	0.2	0.0167	Cycloidal Profile
Profile C	6.5	64.3/64.3	0.2	0.053	3-4-5 Polynomial Profile
Profile D	6.5	64.3/64.3	0.2	0.049	4-5-6-7 Polynomial Profile
Profile E	6.5	64.3/64.3	0.2	0.0124	Splined Profile



(a) Lift of Follower for Various Cams



(b) Velocity V/S Cam Angle



(c) Acceleration V/S Cam Angle

Figure 3: Cam Profile (Lift, Velocity, Acceleration)

Four cam profiles are designed with the values in table 2. Lift of the follower in the modified cam is maintained same as existing one so that the performance of the engine is not affected. Velocity and acceleration for various cams are compared in figure3. Profile C and profile D are not having that much difference in velocity and acceleration. Therefore profile D is not considered for manufacturing. Figure 4 shows the manufactured camshafts.

Figure 4: Camshafts for Testing

4. EXPERIMENTAL SET UP

In order to measure the vibration response and radiated noise from the camshaft assembly, the experiment is conducted with a cylinder head assembly in an open atmospheric condition. Camshaft is driven with help of a 3HP motor. Speed of the motor is adjusted using delta VFD speed controller. Measurement is accomplished using the accelerometer (type 4394 miniature CCLD accelerometer) and microphone. The accelerometer is connected on the crank case to measure

the vibration characteristic between tappet and cam. Microphone is used to measure the sound pressure during opening and closing on the front side at a distance 30cm (Figure 5). A four-channel FFT analyzer is used for data acquisition (make Brüel & Kjær). The investigation focused to measure the sound pressure level and vibration response in the current cam shaft assembly. Further optimization of camprofile is carried out within the ramp area to reduce it. Figure 6 shows the vibration response and sound pressure in the frequency domain when the camshaft is run at 1300rpm. The maximum acceleration is observed 1.2mm/s^2 and overall sound pressure level of 89.86 dB (A).

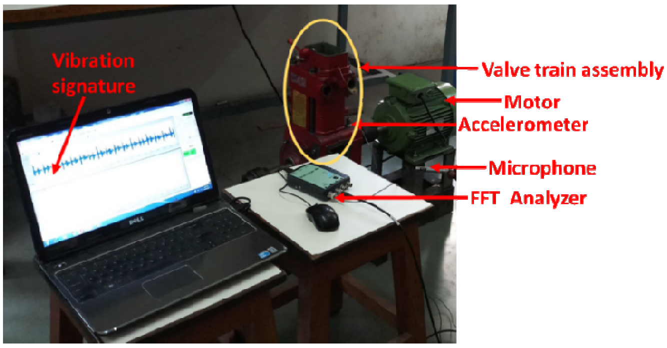
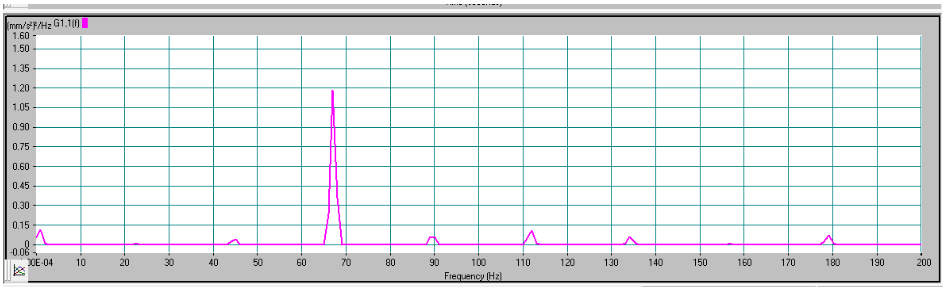
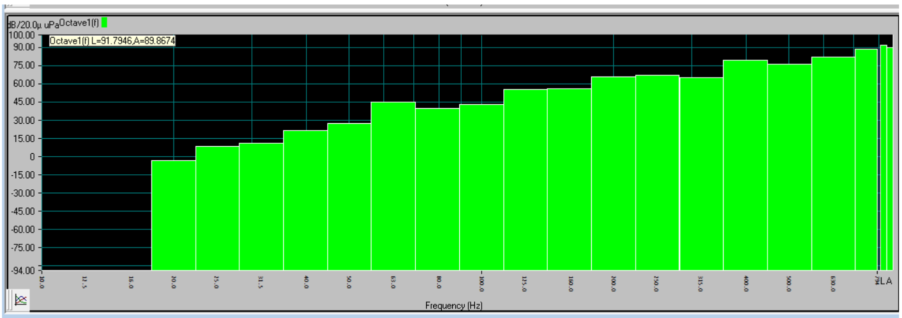


Figure 5: Experimental set up



(a) Vibration



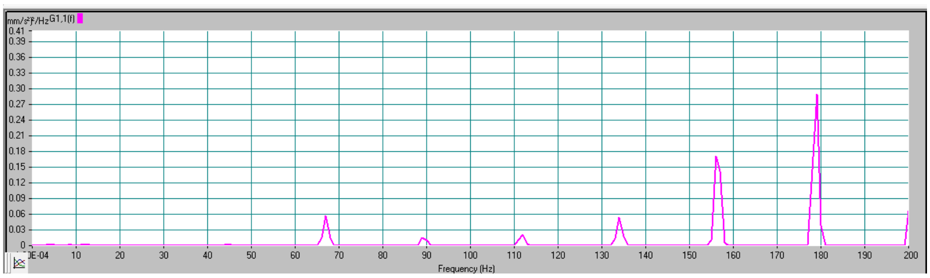
(b) Noise

Figure 6: Vibration and Noise Level in Frequency Domain

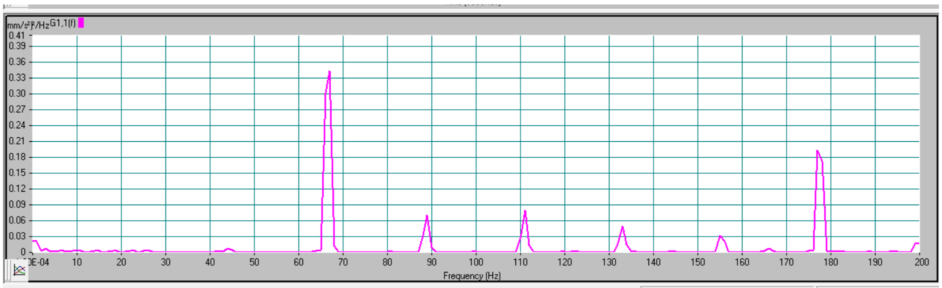
4.1. Vibration Features of Modified Cam Shaft

Experiments have been conducted to measure the vibration response for optimized cam profiles. The vibration level is reduced by the considerable amount in cycloidal cam profile (Figure 7a). There is little increase in the vibration level in other cam profile due to impact between tappet and profile. Similar observations are found in noise level. The radiated noise from the head assembly is reduced by 1.96 dB (A) in case of cycloidal cam profile whereas it is 2.99 dB (A) for splined cam profile (Figure 8). However, the sound quality has changed in all cases. The sharp noise

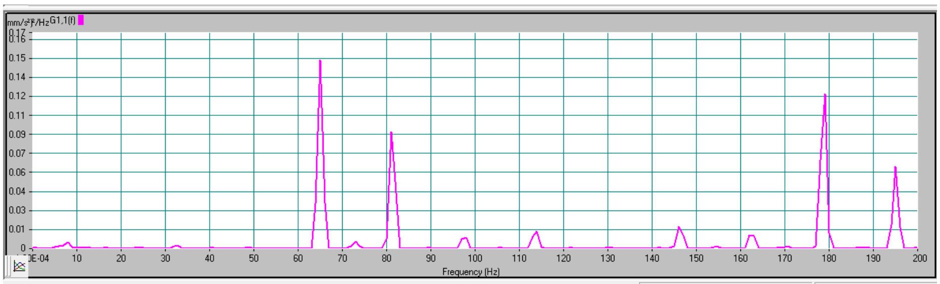
radiated from the head assembly is reduced.



(a) Profile B



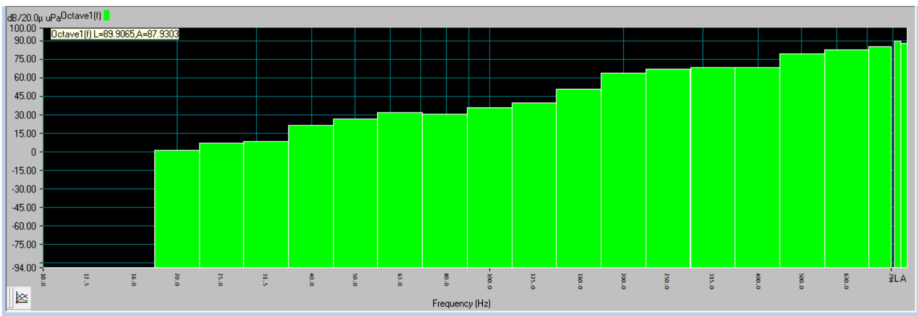
(b) Profile C



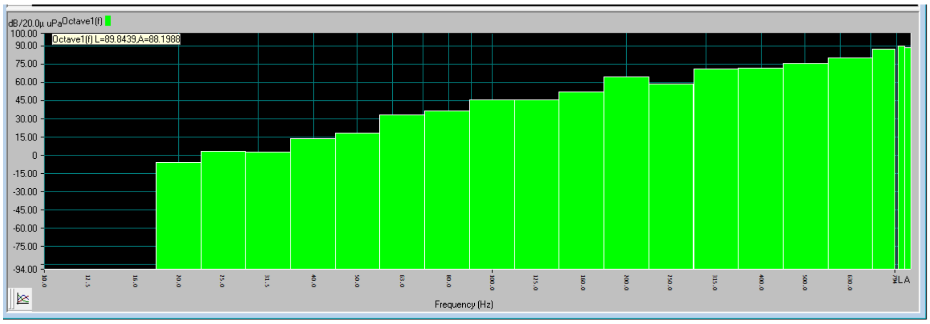
(c) Profile E

Figure 7: Vibration Feature of Optimized Profiles

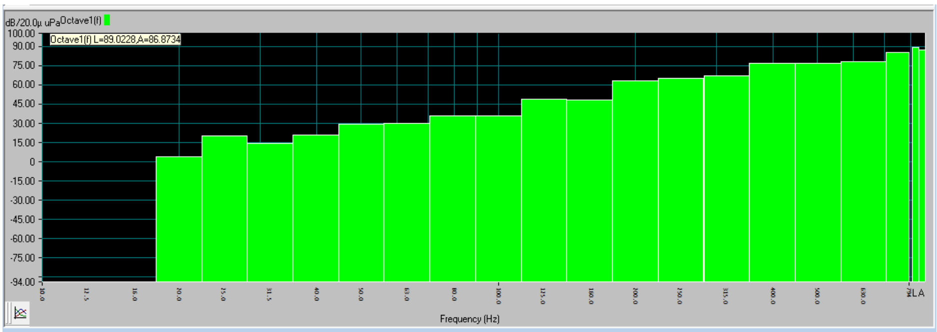
4.2 Noise Features of Modified Cam Shaft



(a) Profile B



(b) Profile C

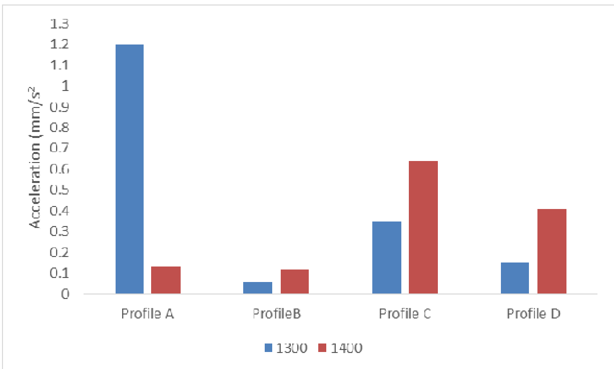


(c) Profile E

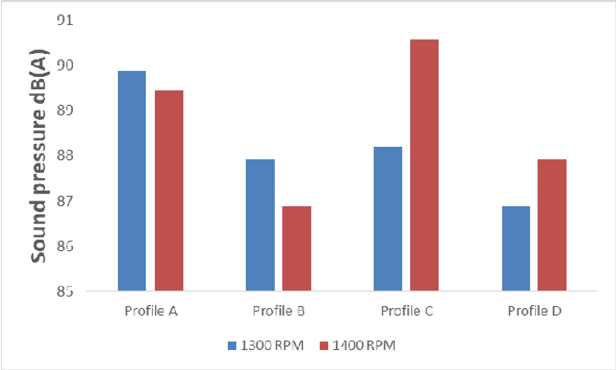
Figure 8: Noise Feature of Optimized Profiles

5. RESULTS AND DISCUSSIONS

From the test results, it has been observed that the vibration response of existing cam is very high as compared to optimized cam profiles (Figure 9a). In profile C and profile E, the vibration response has increased. This is because of impact between tappet and cam. The similar effect is observed in radiated noise from the assembly (Figure 9b). However, the sound quality and sharpness have been improved in modified profiles.



(a) Vibration



(b) Noise

Figure 9: Vibration and Noise Comparison for Various Cam Profiles

6. CONCLUSIONS

This work presents a methodology for cam profile optimization for improvement in Noise and vibration characteristic of valve train assembly. Following conclusions are drawn.

- Due to a sudden increase in velocity of the follower, the vibration level is high in case of a tangent cam.
- By reducing the velocity and acceleration in the ramp area, noise radiated from the valve train assembly can be controlled.
- The vibration response in case of modified cam profile is reduced in magnitude.
- Experimental results show that, the noise radiated from the head assembly is reduced by 1.96 dB (A) for the cycloidal cam. Also, the sound quality characteristics (loudness, impact noise) are smoother in optimized cam profiles.

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